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Combined effect of slip velocity and surface roughness on the ferrofluid based squeeze film lubrication in double layered porous circular plates

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Abstract

Efforts have been made to analyze the combined effect of surface roughness and slip velocity on the ferrofluid squeeze film in double layered porous circular plates. The magnetic fluid flow is governed by Neuringer – Roseinweig model while the stochastic modelling of Christensen and Tonder has been adopted to evaluate the effect of transverse roughness. The associated stochastically averaged Reynolds' type equation is solved to obtain the pressure distribution leading to the calculation of load carrying capacity. The results presented in the graphical forms establish that the magnetization offers a limited scope in containing the adverse effect of roughness, porosity, and slip velocity. However, the situation improves when negatively skewed roughness occurs. But for any type of improvement in the bearing performance the slip has to be kept at reduced level even if variance (-ve) is involved.

Keywords: Circular plate, roughness, ferrofluid, porosity, slip velocity, Load carrying capacity.

NOMENCLATURE:

a	Radius of the circular plate (mm)
r	Radial coordinate
s	slip parameter
h	Uniform fluid film thickness(mm)
$\dot{h}$	Normal velocity of bearing surface
p	Pressure distribution(N/mm^2)
w	Load carrying capacity (N)
μ	Dynamic viscosity of lubricant (N.S/m^2)
μ_0	Permeability of free space (N/A^2)
$\overline{\mu}$	Magnetic susceptibility of magnetic field
σ	Standard deviation (mm)
α	Variance (mm)
ε	Skewness (mm)
H_1	The thickness of the inner layer of the porous plate (mm)
H_2	The thickness of the outer layer of the porous plate (mm)
ϕ_1	The permeability of inner layer (col^{2}kgm/s^2)
ϕ_2	The permeability of outer layer (col^{2}kgm/s^2)
ψ_1	Porosity of inner layer
ψ_2	Porosity of outer layer
$\overline{w}$	Non dimensional load capacity
s^*	Non dimensional slip velocity
α^*	Non dimensional variance
ε^*	Non dimensional skewness
σ^*	Non dimensional standard deviation

INTRODUCTION:

Ferrofluid is widely used in the technical application, dynamic sealing, damping and heat dissipation. Ferrofluid is also used in biomedical application like hyperthermia, constant enhancement for MRI and magnetic separation of cells. Srinivasan [1] investigated the load capacity and time height relationship for squeeze films between doubled layered porous plates, considering various geometries such as annular, circular, elliptic, rectangular etc. Comparison was made between conventional and double layered porous plates. The results showed that load capacity increases due to doubled layered porous plates. Verma [2] presented an investigation for a double layered porous journal bearing using short bearing approximations. The performance characteristics were found to be improved due to low permeability of the inner porous layer. Rao et al. [3] presented an analysis of journal bearing with double layered porous lubricant film using couple stress and Newtonian fluids. A double layered porous lubricant film configuration with a low permeability porous layer on top of a high permeability bearing adherent porous layer improved the bearing performance.

Patel et al. [4] considered the effect of surface roughness on the behavior of a ferrofluid squeeze film between circular plates with porous matrix of variable thickness. It was shown that by taking suitable thickness ratio and magnetic strength the adverse effect of roughness could be minimized, in the case of negatively skewed roughness. Patel and Deheri [5] studied the combined effect of roughness and slip velocity on the Jenkins model Based ferrofluid lubrication of a curved rough annular squeeze film. When the slip was at minimum, Jenkins model based ferrofluid lubrication offered a method for minimizing the adverse effect of roughness considering suitable values of curvature parameter. Cusano [6] conducted the analytical investigation of an infinitely long two layered porous bearing. Results were analyzed for relating the eccentricity ratio and coefficient friction as function of load. Patel and Deheri [7] analyzed the performance of a magnetic fluid based double layered rough porous slider bearing considering the combined porous structures of Kozeny Carman and Irmay. The Kozeny Carman model was found to perform better. Vadher et al. [12] analyzed the behavior of hydromagnetic squeeze film between two conducting rough porous circular plates, it was shown that hydromagnetization compensated the adverse effect of transverse surface roughness to a large extent with the choice of suitable plate conductivities.

Patel and Deheri [14] discussed the combined effect of surface roughness and slip velocity on the Jenkins model based magnetic squeeze film in curved rough circular plates the Jenkins model modified the performance as compared to the Neuringer – Roseinweig model but this model provided little support to negatively skewed roughness to augment bearing performance. Prakash and VIJ [15] analyzed the effect of shape of the plate and porosity on the performance of squeeze films between porous plates of various shapes.

ANALYSIS:

The geometry and configuration of the bearing system is presented below.

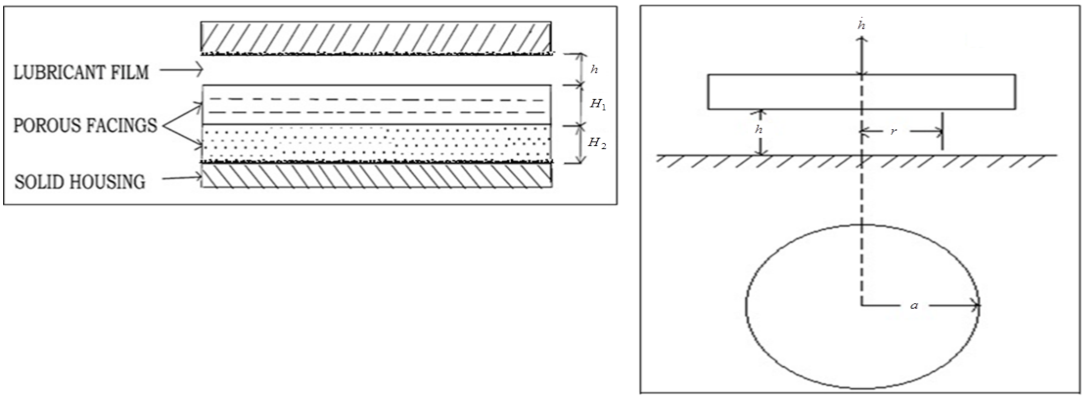

Figure-A: configuration of the bearing system.

The porous regions are assumed to be homogeneous and isotropic. The lubricant is an incompressible Newtonian fluid. The slip model of Beavers and Joseph [16] was used.

The magnetic fluid flow model of Neuringer – Roseinweig consists of following equations.

$$\rho(\bar{q}.\nabla)\bar{q} = -\nabla p + \eta\nabla^2\bar{q} + \mu_0(\bar{M}.\nabla)\bar{H} \tag{1}$$

$$\nabla.\,\bar{q} = 0 \tag{2}$$

$$\begin{aligned} &\nabla\times\bar{H} = 0 \\ &\bar{M} = \bar{\mu}\bar{H} \end{aligned} \qquad (3), (4)$$

$$\nabla.(\bar{H}\times\bar{M}) = 0 \tag{5}$$

Using equations (3) and (4) equation (1) resumes the form

$$\rho(\bar{q}.\nabla)\bar{q} = -\nabla\left(p - \frac{\mu_0\,\bar{\mu}\,H^2}{2}\right) + \eta\nabla^2\bar{q} + \eta\nabla^2\bar{q} \tag{6}$$

In view of the Neuringer - Roseinweig model for magnetic fluid flow [13] and the stochastic averaging model of Christensen and Tonder [9], [10], [11] with the usual assumptions of hydromantic lubrication the generalized Reynolds' type equation for the pressure distribution is given by

$$\frac{1}{r}\frac{d}{dr}\left(r\frac{d}{dr}\left(p-0.5\mu_0\bar{\mu}H^2\right)\right)=\frac{12\mu\dot{h}}{g(h)} \tag{7}$$

Where

$$g(h)=h^3\left(\frac{4+sh}{2+sh}\right)+4\sigma^2h\left(\frac{4+sh}{2+sh}\right)^{\frac{1}{3}}+3\alpha^2h\left(\frac{4+sh}{2+sh}\right)^{\frac{1}{3}}+2\alpha h^2\left(\frac{4+sh}{2+sh}\right)^{\frac{2}{3}}+4\sigma^2\alpha+\alpha^3+\varepsilon+12\phi_1H_1+12\phi_2H_2$$

where in

$S=\frac{k}{\sqrt{\phi_1+\phi_2}}$ is the slip parameter, k is a slip coefficient

And magnitude of the magnetic field is

$$H^2=Ka(a-r),\ 0<r<a \tag{8}$$

Where in K is a suitable constant depending on the material to produce a field of desired magnetic strength.

The associated boundary conditions are

$$p(a)=0\,and\ \left(\frac{dp}{dr}\right)_{r=0}=0 \tag{9}$$

Now integrating equation (7) and using the boundary conditions (9) one gets the expression for the pressure distribution as

$$p=\frac{3\mu\dot{h}}{g(h)}\left(r^2-a^2\right)+0.5\mu_0\bar{\mu}H^2 \tag{10}$$

Now load carrying capacity is calculated from

$$w=\int_0^a rp\,dr$$

which leads to

$$w=\frac{-3\mu\dot{h}a^4}{4g(h)}+\frac{0.5\mu_0\bar{\mu}Ka^4}{6} \tag{11}$$

Now, non-dimensional load carrying capacity is obtained as

$$\bar{w}=-\frac{h^3w}{\mu\dot{h}a^4}$$

$$\overline{w}=\frac{3}{4G(h)}+\frac{0.5\mu^{*}}{6} \tag{12}$$

where

$$\mu^{*}=\frac{-\mu_{0}\,\overline{\mu}\,h^{3}K}{\mu\dot{h}}$$

and

$$G(h)=\left(\frac{4+s^{*}}{2+s^{*}}\right)+4\sigma^{*2}\left(\frac{4+s^{*}}{2+s^{*}}\right)^{\frac{1}{3}}+3\alpha^{*2}\left(\frac{4+s^{*}}{2+s^{*}}\right)^{\frac{1}{3}}+2\alpha^{*}\left(\frac{4+s^{*}}{2+s^{*}}\right)^{\frac{2}{3}}+4\sigma^{*2}\alpha^{*}+\alpha^{*3}+\varepsilon^{*}+12\psi_{1}+12\psi_{2}$$

Where

$$s^{*}=sh,\ \alpha^{*}=\frac{\alpha}{h},\ \varepsilon^{*}=\frac{\varepsilon}{h^{3}},\ \sigma^{*}=\frac{\sigma}{h},\psi_{1}=\frac{\phi_{1}H_{1}}{h^{3}},\psi_{2}=\frac{\phi_{2}H_{2}}{h^{3}}$$

RESULTS AND DISCUSSIONS

As the expression for the load carrying capacity from equation (12) is linear with respect to the magnetization parameter, it can be easily seen that the load carrying capacity will be increased with increasing values of magnetization. Equation (12) suggests that the load carrying capacity increases by 0.08333 μ^{*} as compared to the conventional lubricant based such bearing system.

From figures (1 – 3) it is clearly seen that the load carrying capacity rises sharply with an increase in magnetization. This is due to the fact that the magnetization increases the viscosity of the lubricant. However, initial effect of porosity remains negligible. The fact that bearing suffers heavily because of slip velocity can be seen from figures (4 – 7). Also from figures (5) and (6) it is observed that the load carrying capacity increases as variance (-ve) increases, while, skewness follows the path of variance in this matter. Further, from figure (7) it is found that initial effect of second porous layer is almost nominal.

The load carrying capacity considerably falls because of increasing standard deviation which can be had from figures (8) and (9). Also the initial effect of second porous layer is negligible (figure (9)). Figures (10) and (11) indicate that the combined positive effect of negatively skewed roughness and variance (-ve) may be used for developing a bearing system with enhanced performance. Here also the initial effect of second porous layer remains negligible. The initial effect of first porous layer on load carrying capacity with respect to skewness is negligible (figure (12)).

Some of the figures send the message that with a suitable magnetic strength the combined positive effect of negatively skewed roughness and variance (-ve) may be considered for neutralizing the adverse effect of porosity and slip velocity.

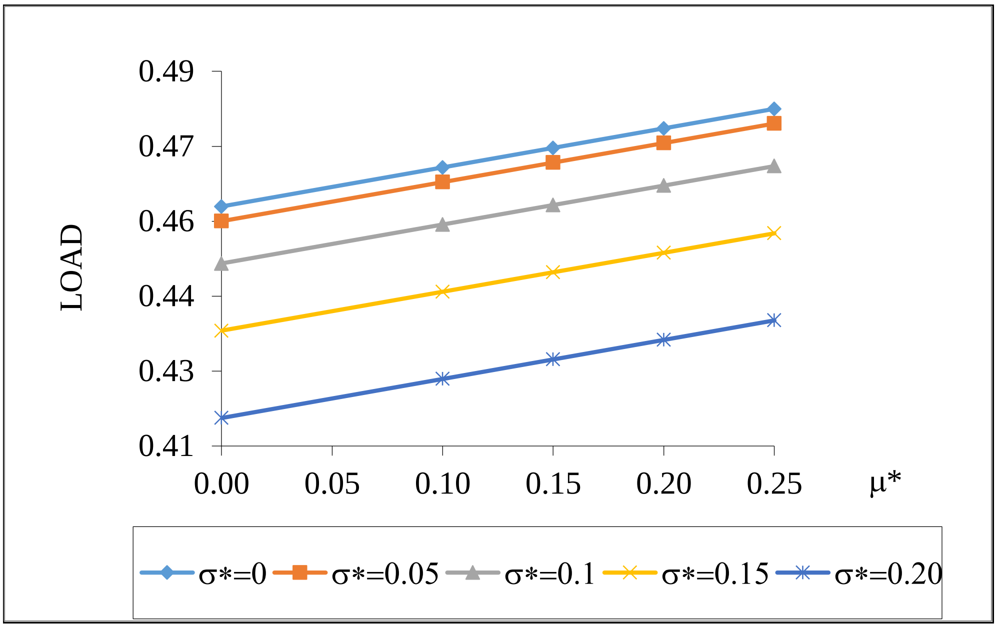

Figure -1 Variation of load carrying capacity with respect to μ* and σ*

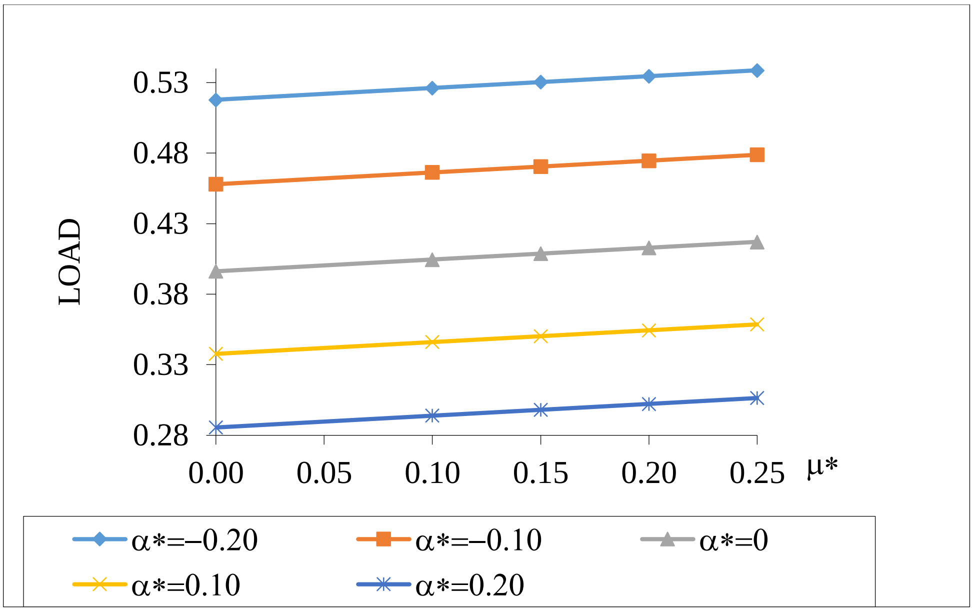

Figure -2 Variation of load carrying capacity with respect to μ* and α*

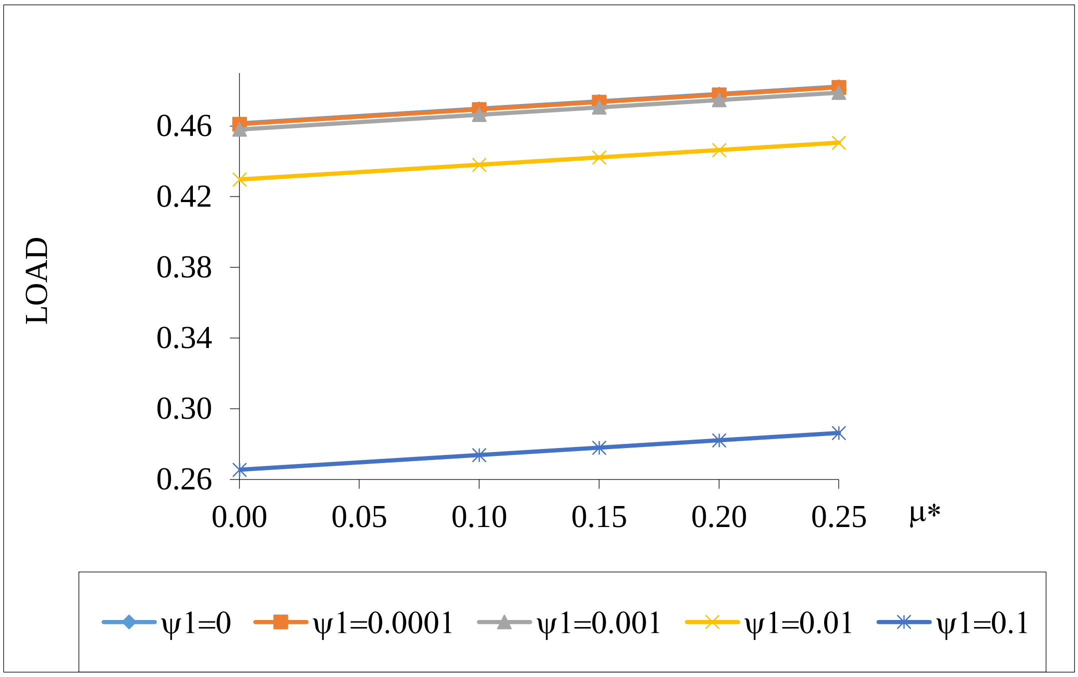

Figure -3 Variation of load carrying capacity with respect to μ* and ψ_1

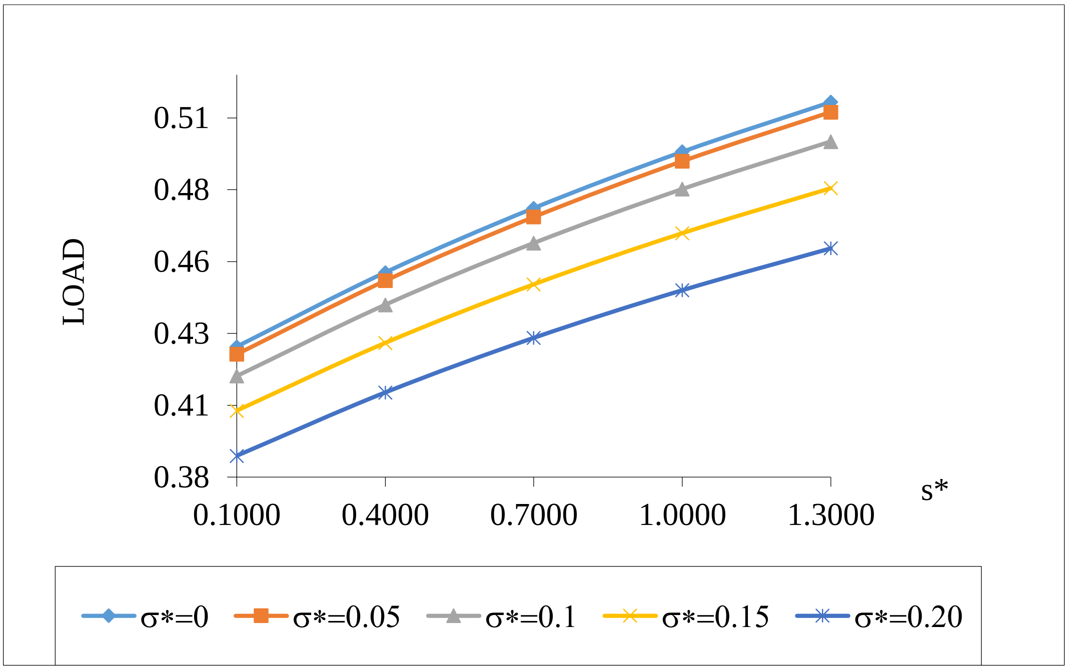

Figure -4 Variation of load carrying capacity with respect to s* and σ*

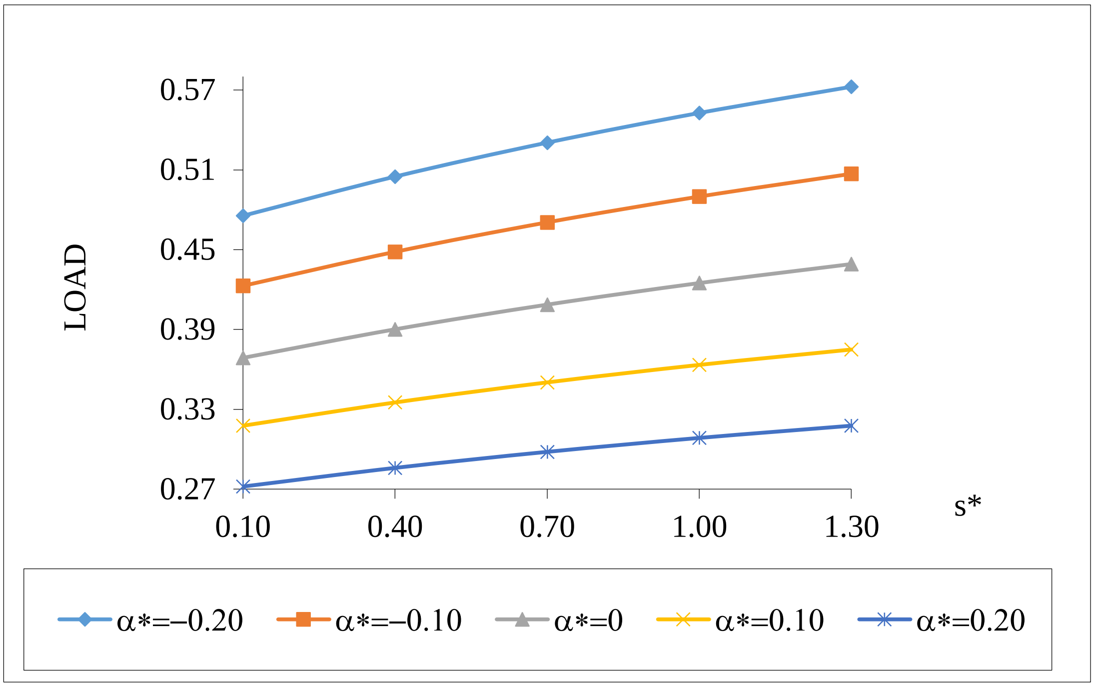

Figure -5 Variation of load carrying capacity with respect to s* and α*

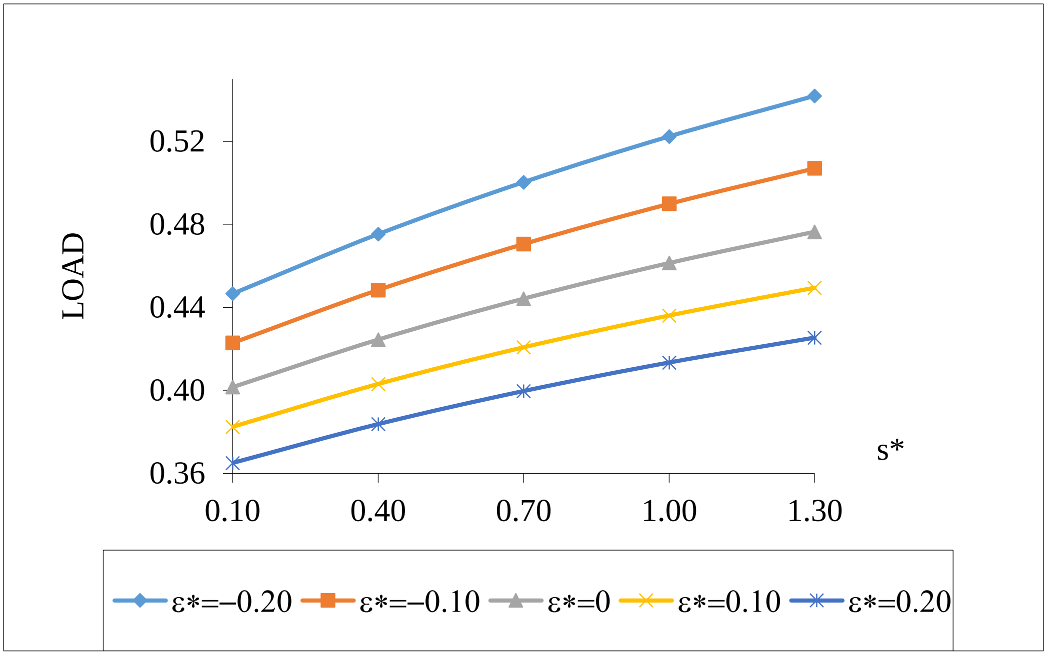

Figure -6 Variation of load carrying capacity with respect to s* and ε*

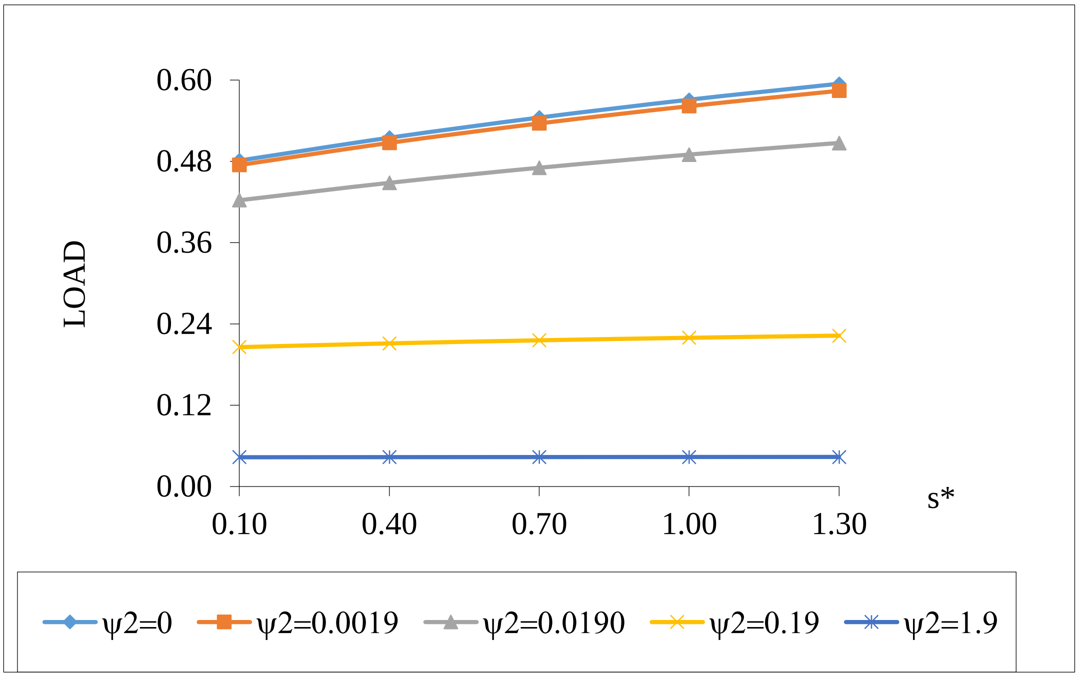

Figure -7 Variation of load carrying capacity with respect to s* and ψ2

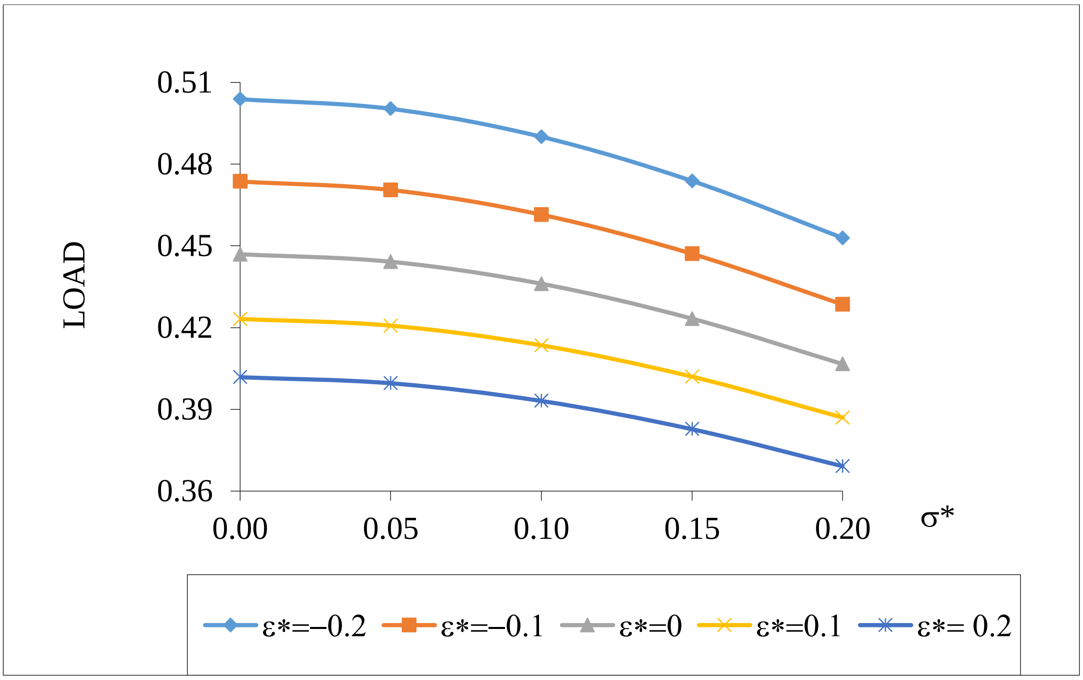

Figure -8 Variation of load carrying capacity with respect to σ* and ε*

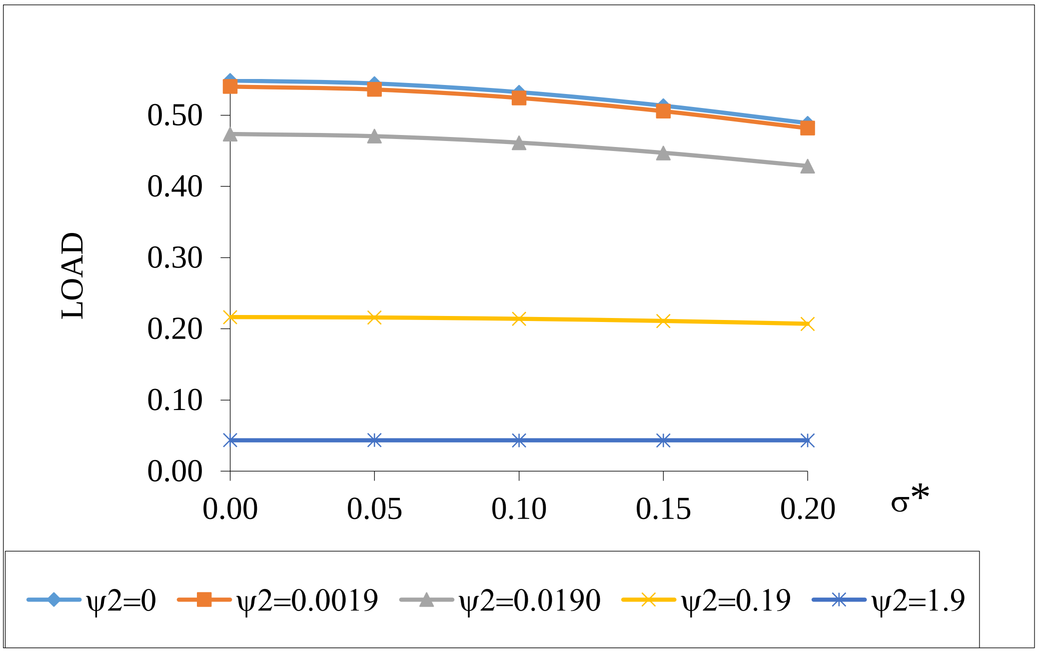

Figure -9 Variation of load carrying capacity with respect to σ* and ψ_2

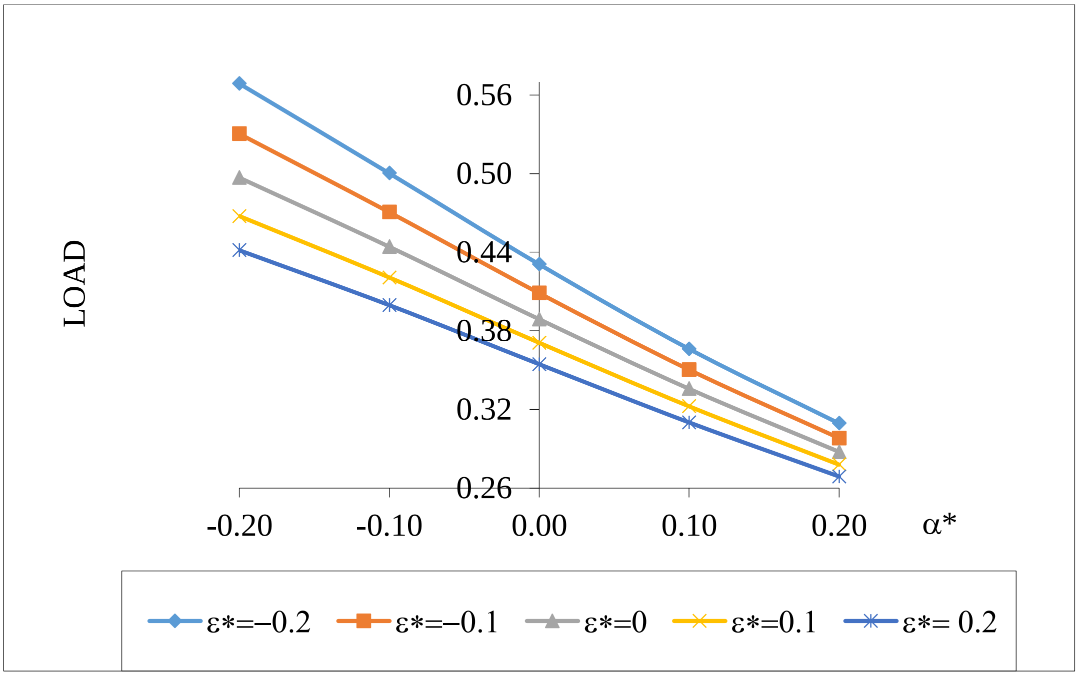

Figure -10 Variation of load carrying capacity with respect to α* and ε*

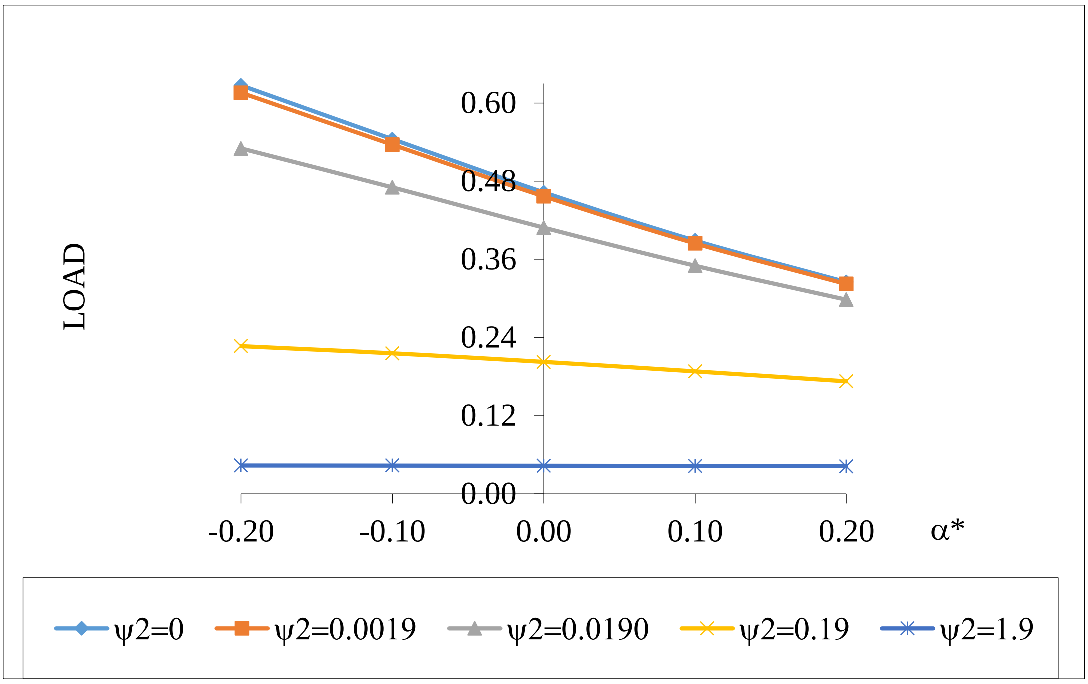

Figure -11 Variation of load carrying capacity with respect to α* and ψ_2

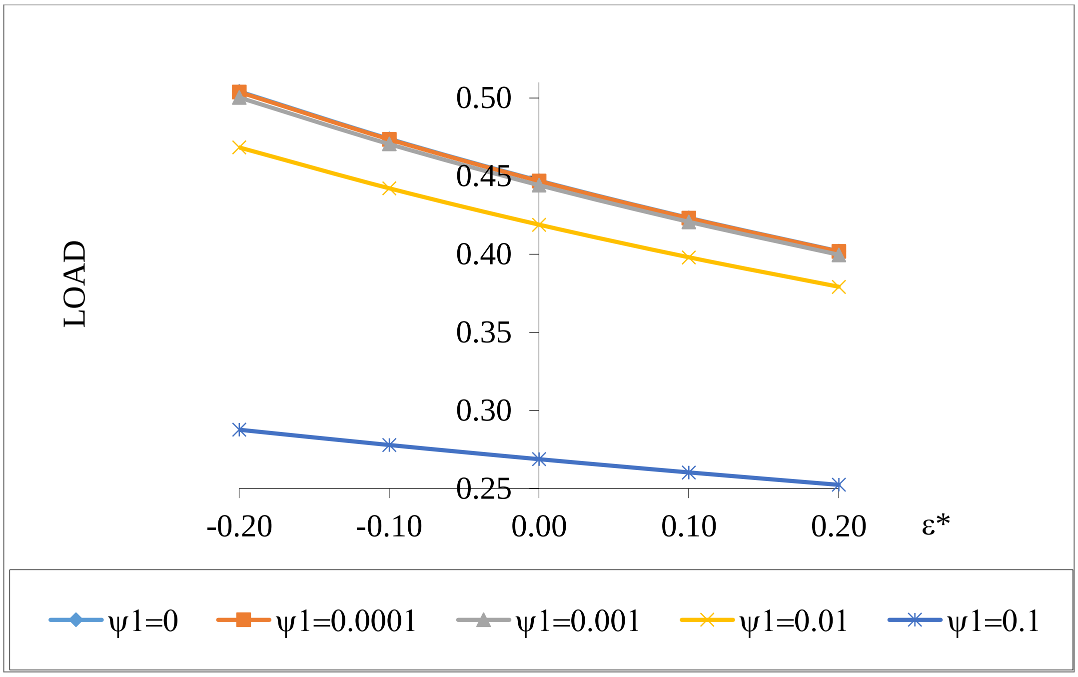

Figure -12 Variation of load carrying capacity with respect to ε* and ψ_1

CONCLUSIONS

1. This investigation makes it clear that for a better performance the slip velocity is required kept at minimum.
2. From design point of view the roughness aspects must be addressed carefully while designing the bearing system.
3. The magnetization offers a limited scope in reducing the adverse effect of roughness, porosity combine even if the slip is at the reduced level. In spite of the adverse influence of many parameters the bearing system supports a good amount of load even when there is no flow, which does not happen in the case of conventional lubrication.

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